United States Patent [19]

Stark

[11] **4,151,430**

[45] **Apr. 24, 1979**

[54] **DYNAMOELECTRIC MACHINE STRUCTURES**

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[73] Assignee: **General Electric Company**, Fort Wayne, Ind.

[21] Appl. No.: **733,415**

[22] Filed: **Oct. 18, 1976**

Related U.S. Application Data

[62] Division of Ser. No. 554,454, Mar. 3, 1975, Pat. No. 3,990,191.

[51] **Int. Cl.**[2] **H02K 15/14**; F16C 1/00

[52] **U.S. Cl.** ..... **310/90**; 310/42; 64/1 V; 306/237 A; 308/DIG. 13

[58] **Field of Search** ..... 310/51, 75 R, 90, 261, 310/262, 42; 64/1 V; 308/216, 237 A, 207 R, 180 R, DIG. 13; 74/574

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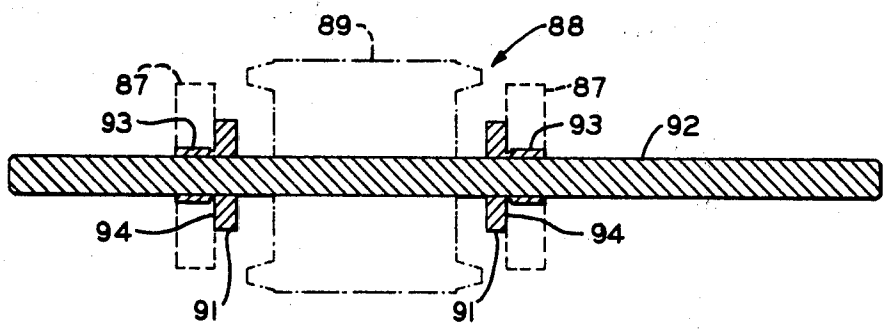
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[57] **ABSTRACT**

A shaft of selected material and of a given desired nominal diameter corresponding to nominal desired shaft extension diameter for a particular motor is cut and then treated with optional operations such as chamfering and washing; grinding; machining of flats; and phosphatizing to inhibit rust. A first accessory member is assembled in a predetermined position by heat shrinking on the shaft; and a rotor body is assembled with the shaft. A second accessory member is also assembled by heat shrinking. For ball bearing applications, concenntric journals are turned on accessory members for the inner races of bearings after the accessory members have been assembled with shaft. The inner races of ball bearings are lightly pressed onto the journals, and rotors, end frames, and stator are assembled. The concentric journals permit press fitting of ball bearings without substantially exceeding 0.0005 of an inch press fit and thus avoid damage to the bearings.

Shaft portions are stressed and maintained in a stressed condition so as to increase the stiffness of the rotor system and thereby increase the rotor system resonant frequency. Aluminum collars that conform to the shaft surface are preferably used. These collars establish a more thorough and uniformly stressed condition of the shaft.

14 Claims, 15 Drawing Figures



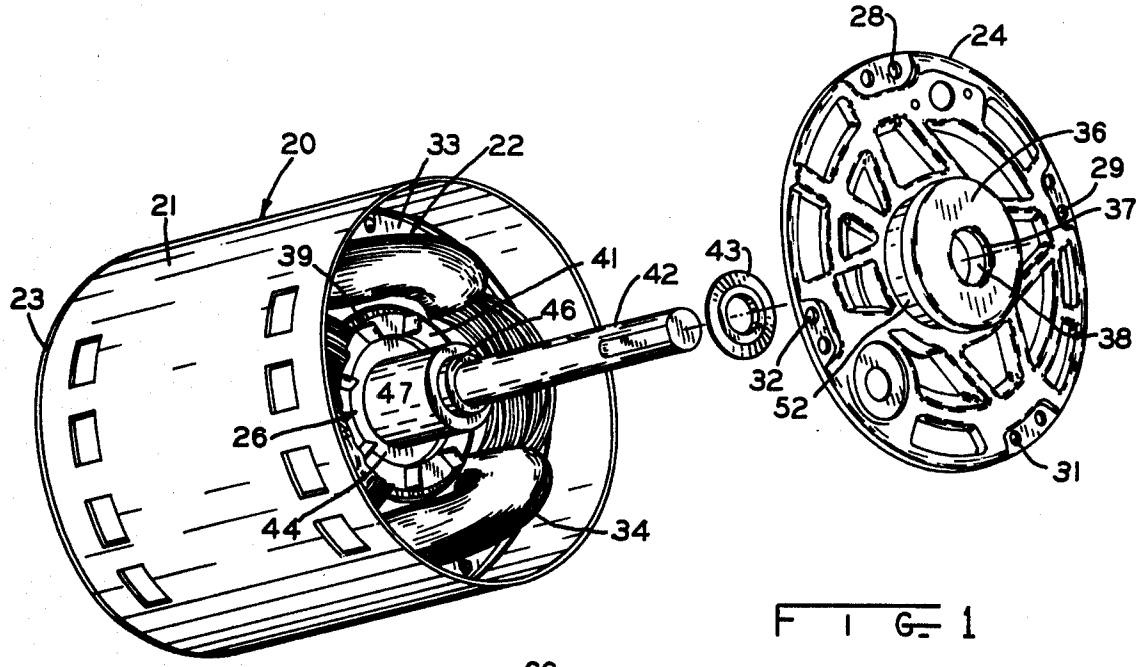


FIG. 1

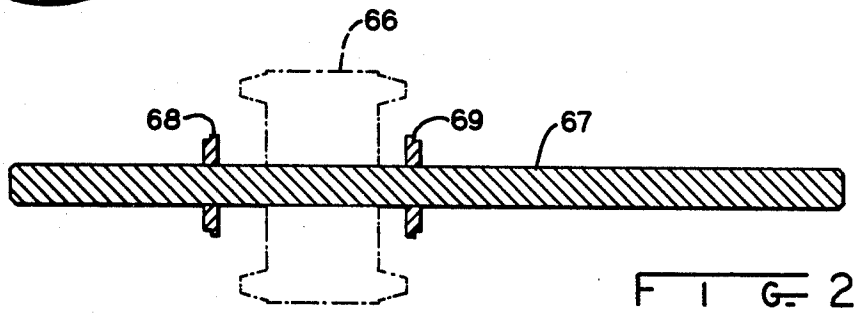


FIG. 2
PRIOR ART

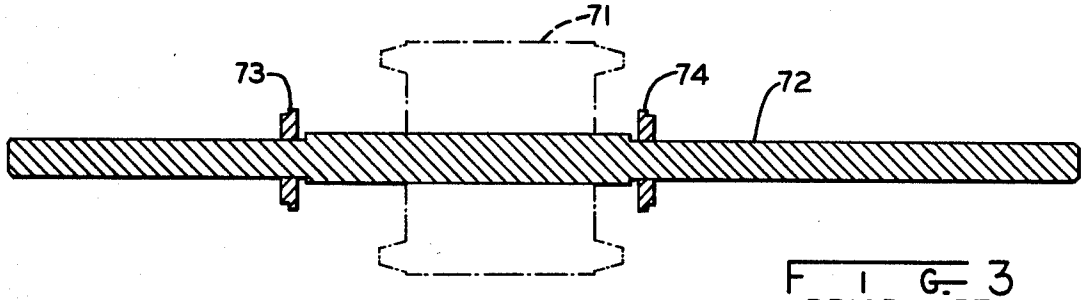


FIG. 3
PRIOR ART

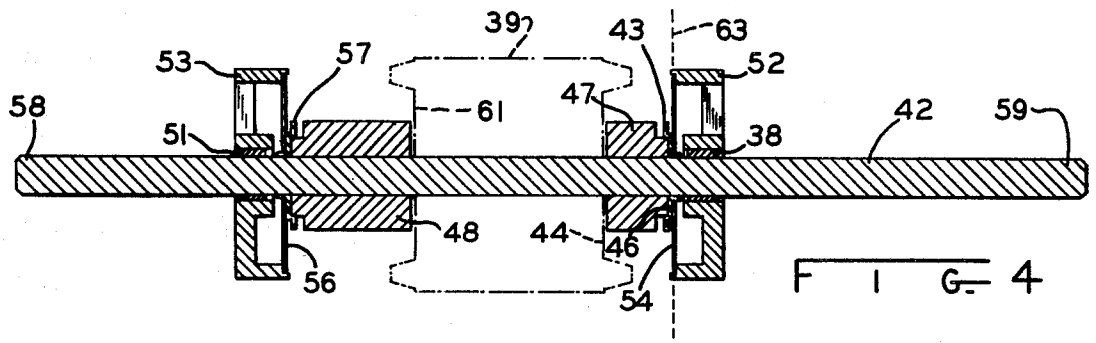


FIG. 4

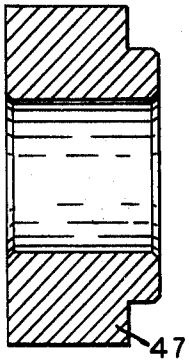


FIG. 5

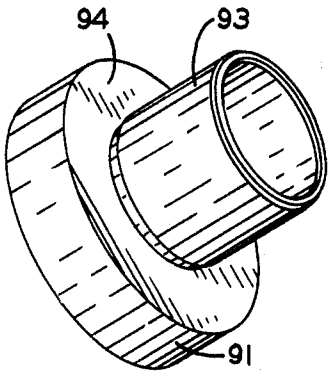


FIG. 9

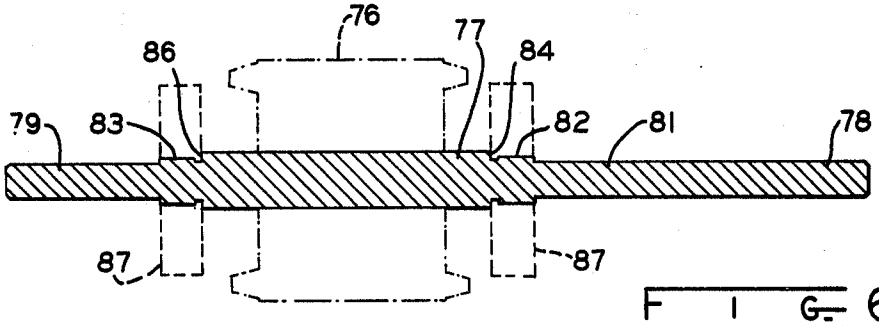


FIG. 6
PRIOR ART

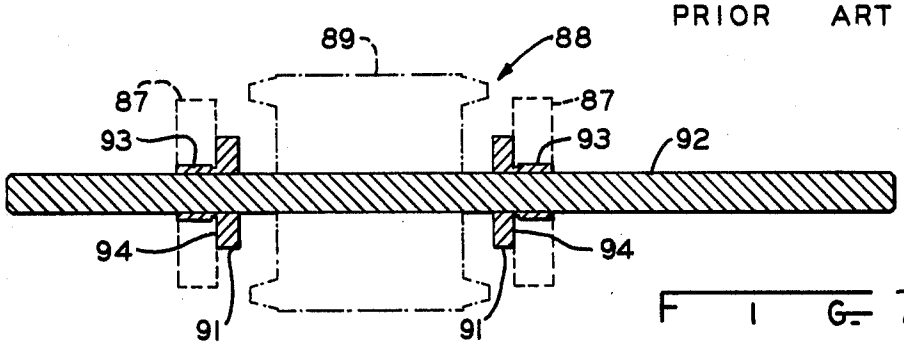


FIG. 7

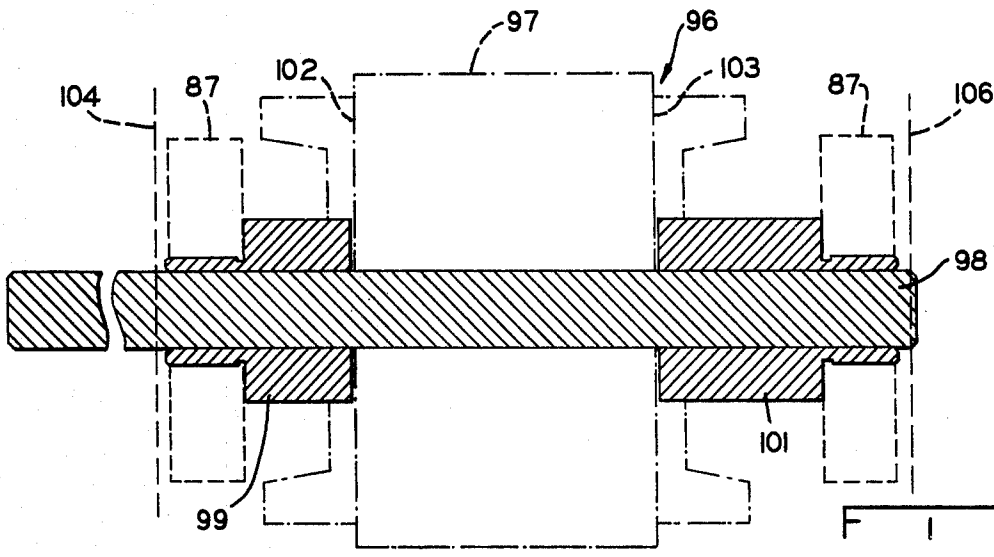


FIG. 8

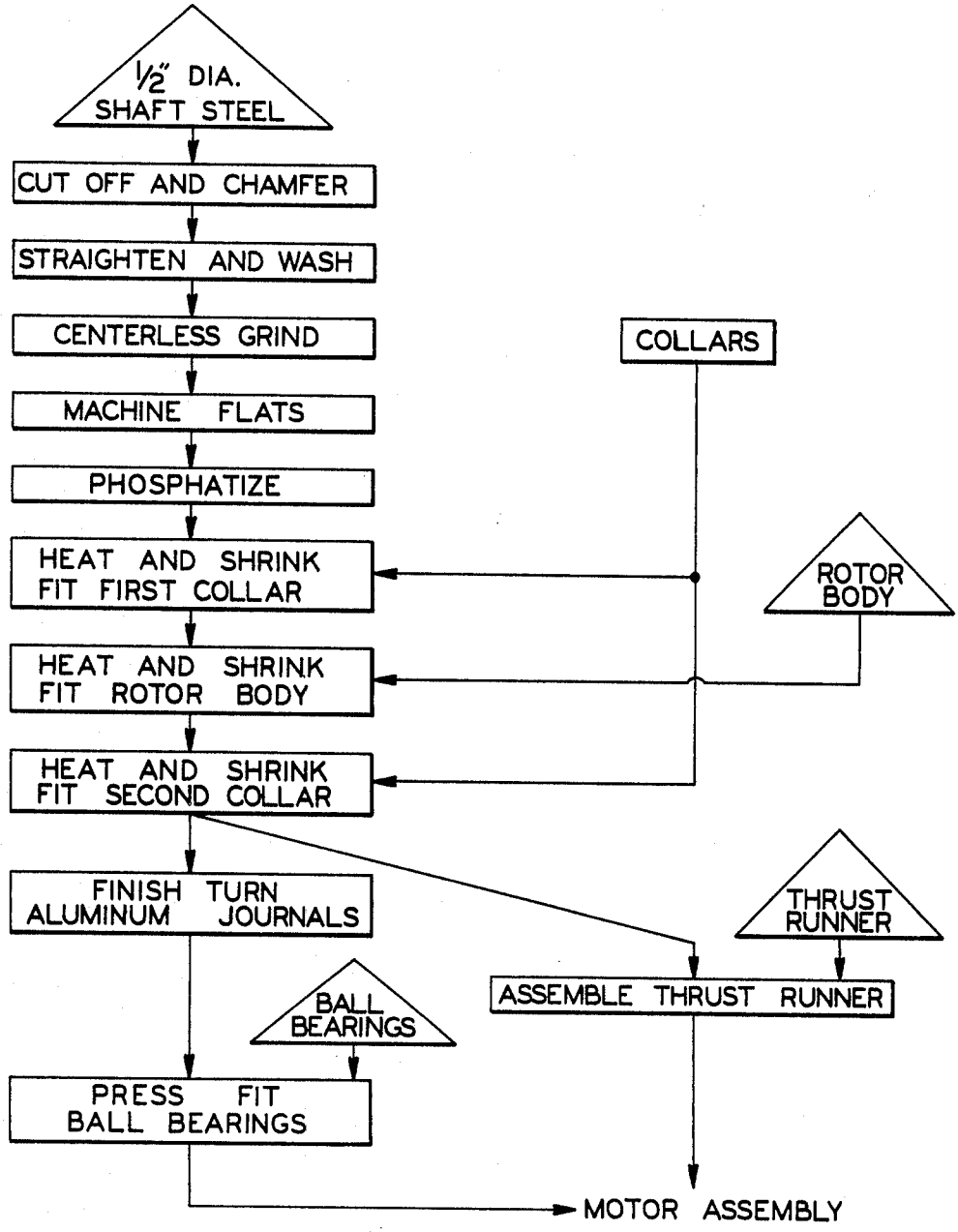

1/2" DIA. SHAFT STEEL
CUT OFF AND CHAMFER
STRAIGHTEN AND WASH
CENTERLESS GRIND
MACHINE FLATS
PHOSPHATIZE
HEAT AND SHRINK FIT FIRST COLLAR
HEAT AND SHRINK FIT ROTOR BODY
HEAT AND SHRINK FIT SECOND COLLAR
FINISH TURN ALUMINUM JOURNALS
PRESS FIT BALL BEARINGS
COLLARS
ROTOR BODY
THRUST RUNNER
ASSEMBLE THRUST RUNNER
BALL BEARINGS
MOTOR ASSEMBLY

FIG 10

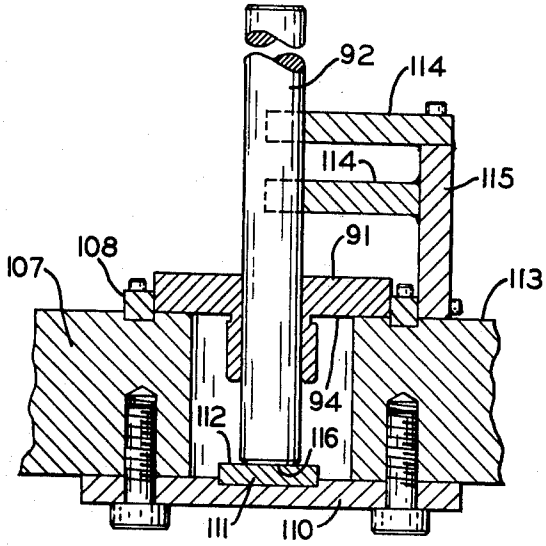


FIG. 11

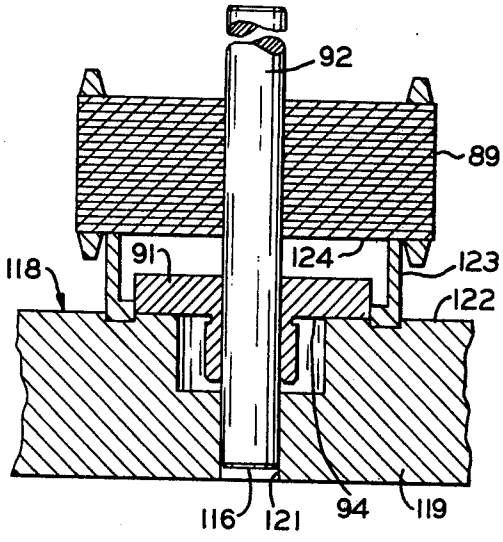


FIG. 12

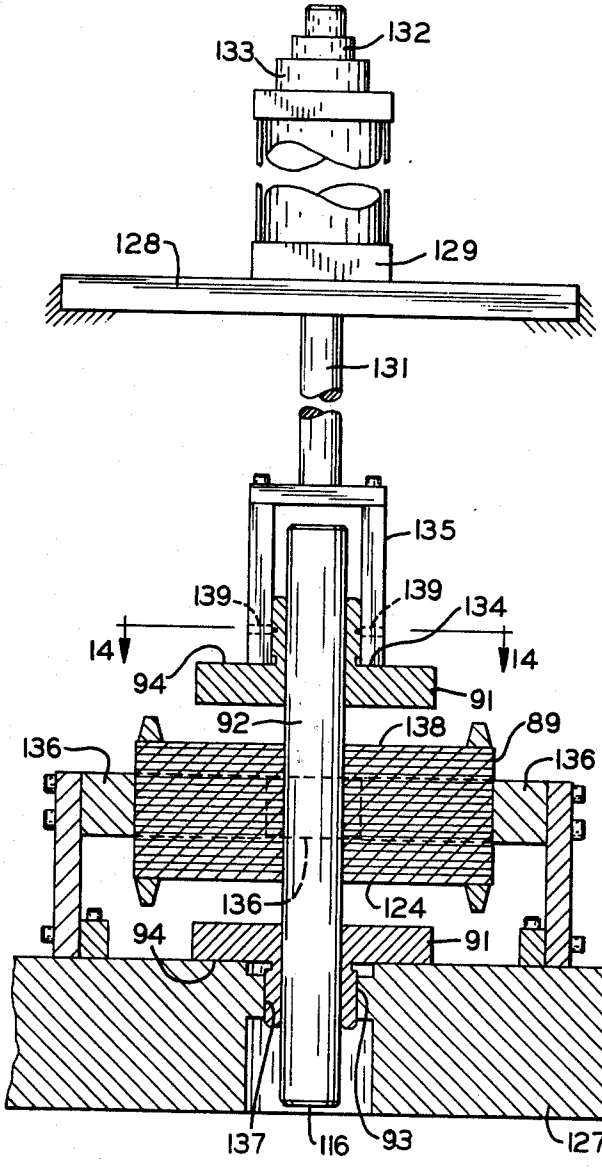


FIG. 13

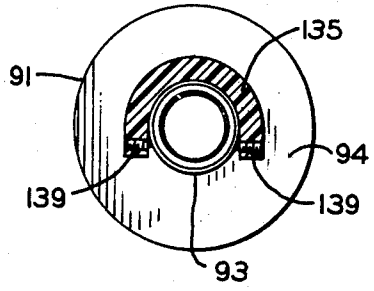


FIG. 14

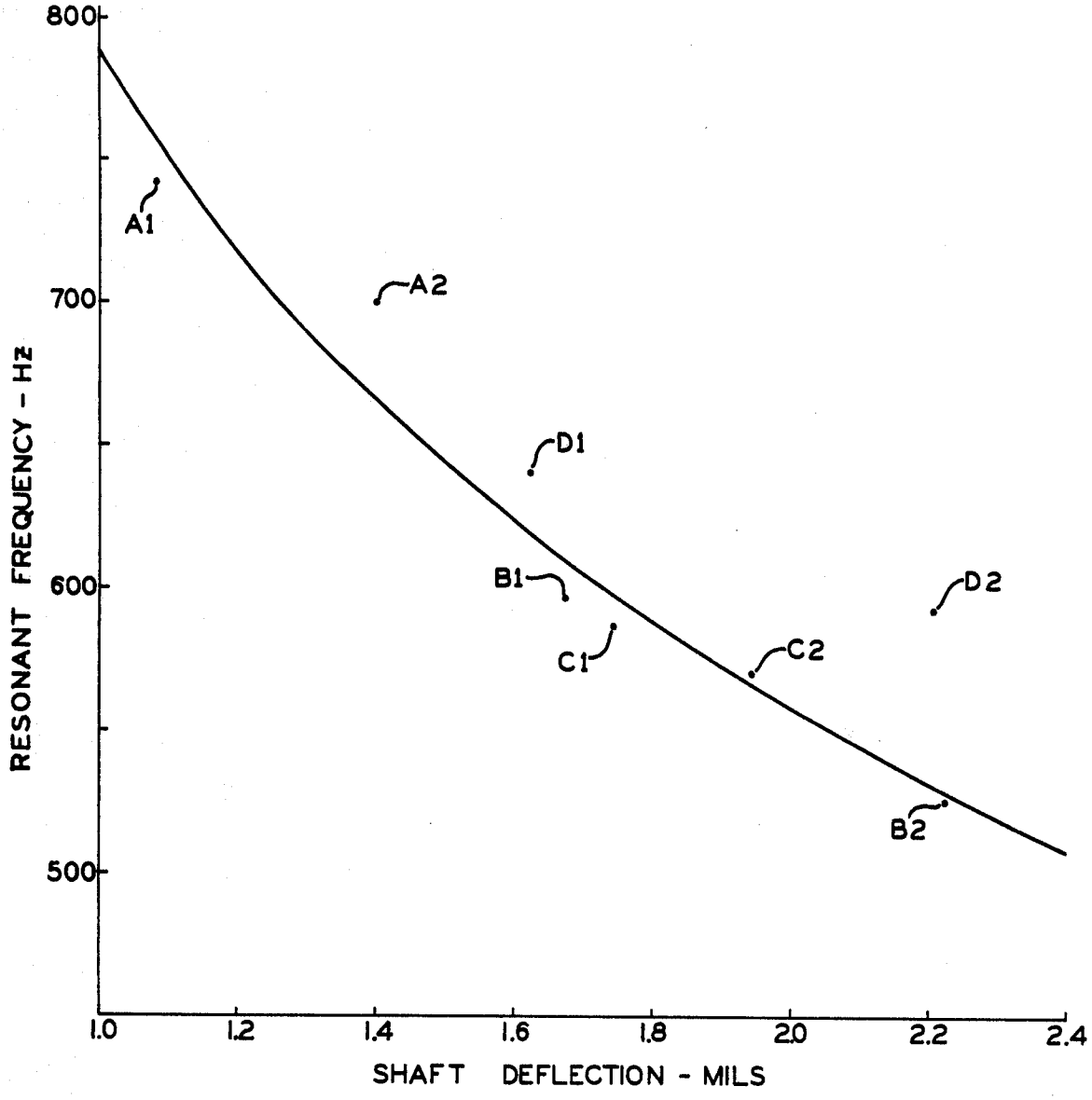

800
700
600
500
RESONANT FREQUENCY - Hz
A1
A2
D1
B1
C1
C2
D2
B2
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
SHAFT DEFLECTION - MILS

FIG. 15

DYNAMOELECTRIC MACHINE STRUCTURES

CROSS REFERENCE TO RELATED APPLICATION

This application is a division of my now co-pending application Ser. No. 554,454 which was filed Mar. 3, 1975 and which issued Nov. 9, 1976 as U.S. Pat. No. 3,990,191. The entire disclosure of said application is incorporated herein by reference.

BACKGROUND OF THE INVENTION

The present invention relates generally to rotatable dynamoelectric machines, and, more particularly, to rotatable assemblies usable therein.

Shafts of rotary dynamoelectric machines, such as fractional horsepower electric motors of both ball and sleeve bearing types, often are machined from shaft stock that initially is of a relatively large diameter. Extensive machining of large diameter shaft stock often has been required, even though one or more external shaft extensions are required to be of a small diameter. In some cases, large diameter stock is used so that intermediate diameter seats, and large diameter shoulders may be provided thereon for ball bearings. Moreover, in the case of shafts designed to accommodate ball bearings, a groove may be required adjacent the ball bearing seat to provide clearance for the inner race of the bearing.

Some sleeve bearing motors are provided with straight shafts, i.e., shoulders, steps, etc. are not normally provided. However, for other sleeve bearing motors, it has sometimes been necessary to provide enlarged shafts in order to increase the stiffness thereof, and machine the shaft extensions to a smaller, acceptable size. This usually is done in order to avoid vibration and noise problems when either the horsepower of a motor or the distance between bearings is increased.

Although the above problems may be encountered with different motors having different diameter shaft extensions, the problem as it pertains to motors having one-half inch (12.7 mm) diameter shaft extensions will now be set forth in more detail.

For some ball bearing motors with a one-half inch diameter shaft extension, it has been necessary to machine the shaft from three-quarter inch (19 mm) diameter stock. In these cases, multiple machining operations have been performed on each end of the shaft to provide a central shaft portion that remained three-quarter inches (19 mm) in diameter, machined seats for a ball bearing [e.g., for a ball bearing with an inner race diameter of about 0.590 inch (15 mm)]; and one-half inch (12.7 mm) shaft extensions.

On the other hand, for some sleeve bearing type motors that required a nominal one-half inch (12.7 mm) shaft extension, one-half inch diameter ends have been machined on a five-eights inch (15.9 mm) diameter shaft to provide bearing journals and one or more shaft extensions of the desired one-half inch (12.7 mm) size. Five-eights inch (15.9 mm) diameter shaft material has been used so that the unsupported length of the shaft between the bearings would be stiffer and have less tendency to bend or whip during operation and thus generate less noise (as compared to a one-half inch (12.7 mm) straight shaft). As will be understood, such noise is objectionable because it may be transmitted to the environment directly, or through the structural mechanisms that are interconnected with either the motor shaft or the motor shell. In other one-half inch (12.7 mm) shaft, sleeve bearing motor applications, the axial length of the rotor, and unsupported shaft length have been sufficiently small that one-half inch (12.7 mm) shaft material has been used with no particular problem.

In large volume production operations, the need to produce different shafts because of different rotor lengths or bearing types increases manufacturing costs not only due to extra material usage and machining operations; but also because of extra capital investment for machining equipment and because of the expense associated with maintaining different types of production equipment.

Still another expense associated with the use of differently sized shafts is more directly related to the manufacture of the rotor body. It should be recalled that squirrel cage rotors for induction motors are made up of laminations that have a centrally located bore for accommodating the shaft. All of the above specifically described motors typically would have rotors with a common outer diameter. However, laminations for each different motor would have a shaft receiving bore of one-half inch (12.7 mm), five-eights inch (15.9 mm), or three-quarters inch (19 mm), depending on the diameter of the central portion of the shaft.

It should now be understood that it would be extremely desirable to provide new and improved rotor arrangements that can be utilized to overcome any one or more of the above-mentioned problems.

Accordingly, it is a general object of the present invention to provide improved rotor arrangements, as well as dynamoelectric machines including such rotor arrangements, so that the above-identified and other problems may be overcome.

It is a more specific object of the present invention to provide a new and improved rotor assembly whereby motors that require a given single diameter shaft extension may utilize straight shafts of a minimum diameter, even for ball bearing or long rotor applications that have previously required stepped or otherwise machined shafts.

It is a more specific object of the present invention to provide a new and improved rotor assembly wherein accessory means, assembled on a straight shaft so as to form part of the rotor assembly, satisfy the previous need for stepped shafts.

SUMMARY OF THE INVENTION

In carrying out the above and other objects of the present invention in one form thereof, shaft material of a given desired nominal diameter corresponding to the nominal desired shaft extension diameter for a particular motor is selected.

Thereafter, shaft blanks are cut and, when desired, optional operations such as chamfering and washing of the shaft blank are performed. The shaft blank may then be ground, at least one flat may be machined thereon, and a rust inhibiting treatment (e.g., phosphatizing) is performed. A first accessory member is assembled in a predetermined position on the shaft (preferably by heat shrinking), a rotor body is assembled with the shaft, and a second accessory member is assembled on the shaft (again, preferably, by heat shrinking).

If the rotor assembly is to be utilized in a sleeve bearing application, thrust members preferably are assembled with the shaft, and thereafter the rotor assembly, stator, and end frames of the motor are assembled. On the other hand, if the rotor is to be utilized in a ball

bearing application, concentric journals are finish turned on the accessory members for the inner races of such bearings after the second accessory member has been assembled with the shaft. The inner races of the ball bearings are pressed onto the journals and the rotor is assembled with parts such as end frames and a stator to form a complete motor. The concentric journals permit ball bearings to be press fit thereon without substantially exceeding a 0.0005 of an inch press fit and thus causing damage to the bearing units.

In the case of rotor assemblies that are to be used for sleeve bearing motors (and optionally for some that are to be used for ball bearing motors), it is preferable that the shaft portions which extend from a location proximate to the face of the rotor body to a location proximate to a bearing supported portion thereof are stressed, and this stressed condition of the shaft is maintained. The stressed condition of the shaft is maintained even after assembly of the motor has been completed, and this increases the stiffness of the rotor system and therefore also increases the resonant frequency of the rotor system.

Apparatus for manufacturing assemblies embodying the invention may include means for maintaining a desired perpendicular relationship between a first accessory member and a shaft while the two are being assembled together. The same or other apparatus also preferably includes means for establishing a first dimensional relationship between the face of a rotor body and the first accessory member while the rotor body is being assembled to the shaft. The same or other apparatus also preferably would be utilized to establish and maintain a desired dimensional relationship between the first and second accessory members as the second accessory member is assembled on the shaft.

When carrying out the invention in one preferred form, accessory members are produced from aluminum or aluminum alloy, heated to a temperature in a predetermined temperature range such as 400° to 700° F. (204° C. to 371° C.), slipped onto a steel shaft, and thereafter cooled so that the accessory members will shrink about the shaft as they cool to ambient, and tightly grip and stress the surface of the shaft. It is preferable that a material such as aluminum be selected so that it will conform to the shaft surface and establish a more thorough and uniformly stressed condition of the shaft. However, steel accessory members may be used although my preferred approach provides results that are believed to be generally unexpected and surprisingly advantageous. The auxiliary members extend from a location proximate to the end face of a rotor body, to a position proximate to the portion of the shaft that is to be supported by a bearing. Furthermore, the auxiliary members are each shrunk with a predetermined interference fit, and the members may include a bearing surface. In one form, such surface is a face against which a thrust runner may bear. In another form, such surface is a seat for a ball bearing.

When a ball bearing seat is provided, a ball bearing inner race is pressed onto the seat. In yet another form of the invention, collars are used that provide a shoulder and seat for a ball bearing without stressing the entire unsupported span of shaft between the rotor body and bearing location. When constructions according to the above-described forms are provided, it is preferably to establish sufficiently tight fits that the collars are not dislodged during subsequent handling or manufacturing operations.

BRIEF DESCRIPTION OF THE DRAWINGS

The subject matter which I regard as my invention is particularly pointed out and distinctly claimed in the concluding portion of this specification. The invention itself, however, together with further objects and advantages thereof, may be better understood by reference to the following description taken in conjunction with the accompanying drawings in which like reference numerals have been used to identify the same parts, and in which:

FIG. **1** is a partially exploded perspective view (with parts broken away and other parts removed) of a motor that includes structures embodying the present invention in one form thereof, such structures having been made by using apparatus disclosed herein;

FIGS. **2** and **3** are schematic representations of prior constructions;

FIG. **4** is a view in elevation, with parts removed and parts in phantom, of part of the motor of FIG. **1**;

FIG. **5** is a sectional view (to a somewhat enlarged scale) of one of the parts shown in FIGS. **1** and **4**;

FIG. **6** is a side elevation, with parts in phantom and parts broken away, of another construction that has been used long prior to the present invention;

FIG. **7** is a side elevation, with parts in section, and parts removed, of a modified structure that may be used in motors embodying the invention in another form;

FIG. **8** is a view (to an enlarged scale), of a rotor assembly that embodies the invention in yet another form;

FIG. **9** is an enlarged perspective view of one of the parts illustrated in FIG. **7**;

FIG. **10** is a flow chart of steps that may be followed when making assemblies embodying the present invention in one form thereof;

FIGS. **11**, **12** and **13** are schematic representations of apparatus useful when carrying out the invention;

FIG. **14** is a view taken along the lines **14—14** in FIG. **13**; and

FIG. **15** is a curve which is a plot of resonant frequency versus shaft deflection for prior constructions and for constructions that embody the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

In FIG. **1**, an electric motor **20** has been illustrated to exemplify structures, embodying the invention in one form thereof, that may be made as taught herein, with or without the apparatus disclosed herein.

The motor **20** includes a conventional shell or housing **21** which accommodates a stator assembly **22**, and also includes a pair of end frames or shields **23**, **24**, and a rotor assembly **26**. Means for mounting motor **20** to a base, appliance, or other structure have not been shown, but it will be understood by persons skilled in the art that mounting rings may be placed on a hub **52** carried by each end shield **23**, **24**, and that such rings may then be disposed in any suitable cradle or base member. Alternatively, motor **20** may be provided with axially projecting studs on either of the end frames, or not shown clamp bolts (which pass through end frame holes **28**, **29**, **31**, **32**) may be used for motor mounting purposes. The description presented herein of various approaches for mounting motor **20** is for purposes of description only and any other desired mounting approach may be followed.

Also for purposes of description rather than limitation, it is noted that the stator assembly **22** includes a four-pole shaded salient pole magnetic iron core **33** and four concentrated coil excitation windings **34** (it being noted that one of these has been broken away for purposes of illustration). Moreover, the end frames **23, 24** have been shown as two aluminum die cast members that are identical to one another, even though motor **20** is a single shaft extension motor. Cover **36** is provided with a central shaft admitting aperture **37** but the corresponding cover for end frame **23** is not provided with a similar aperture.

As will be understood, the end frames **23, 24** are each provided with a seat which accommodates a bearing; and the covers **23, 24** in fact each have a sleeve bearing **38** assembled therewith. However, it also will be understood that the bearing hubs of the end frames used in a motor intended for use in a different application may seat the outer race of a ball bearing.

The rotor assembly **26** includes a rotor body **39** which comprises a laminated magnetic core and cast aluminum conductors which are short circuited with one another at each end by cast aluminum end rings **41**, all as will be understood. The rotor body **39** may be secured to a shaft **42** by heat shrinking or by other means, e.g., by an adhesive, by a mechanical press fit, mechanical deformation of end laminations against the shaft, and so forth. Rotor assembly **26** includes a thrust runner **43** which is normally pressed onto shaft **42**, but which is removed therefrom in FIG. **1** for purposes of illustration. Thrust runner **43** is made of graphite impregnated nylon, e.g., "Nylation" as sold by the Polymer Corporation, but any other suitable low friction material may be used.

Also included, as part of rotor assembly **26**, are two accessory members which are heat shrunk onto shaft **42** so as to stress at least that portion of the shaft material along the surface of the shaft. One of these members is shown in FIG. **1** as an aluminum collar **47**. The portion of the shaft **42** that extends from an axial location on the shaft that is proximate to the face **44** of rotor body **39** to an axial location that is proximate to the end face **46** of collar **47** is held continuously in compression during all normal temperature conditions, due to the hoop tension in collar **47**.

Arrangements embodying the invention thus far described provide advantageous results in terms of motor generated noise during operation of motor **20**. These improved results are presently believed to be due to the stressed condition of the shaft (as just described) at each end of rotor body **39**; and also perhaps due to some increase in shaft strength due to the mass of collar **47**. Since these beliefs and present understanding are discussed in more detail hereinbelow, it is only noted further at this point that collars **47** and **48** (See FIG. **4**) were machined from 2011-T3 aluminum, that shaft **42** was made from cold drawn low carbon steel; (e.g., one having an AISI designation of C1211 to C1215) and that collars **47, 48** were heated to a temperature in the predetermined temperature range of 400° F. to 700° F. (204° C. to 375° C.) before they were relatively slipped along shaft **42** (while the shaft was held at room ambient temperature of about 21° C.) to predetermined axial locations, and allowed to shrink against the shaft at such locations.

The degree to which the collars must be heated may vary, depending on the material from which the collars are made, the diameter of the shaft, and the bore diameter of the collars. It is preferable to size the bore of collars (that are to fit on one-half inch or 12.7 mm shafts) so that there would be an interference fit on the shaft of from about 0.0009 inch to 0.0020" (0.0228 mm to 0.0508 mm). More specifically, I have obtained entirely satisfactory results when the collar bore diameter has been machined to a tolerance of 0.4987 inch (12.67 mm) maximum and 0.4980 inch (12.65 mm) minimum; and the shaft diameter has been held to a dimension within the range of from 0.4995 inch (12.69 mm) to 0.5000 inch (12.7 mm).

Exact dimensions have now been described for the purpose of providing a full and complete teaching, but it should be understood that departures could be made therefrom while still utilizing the invention. When the various parts are made of the materials described hereinabove and dimensioned as noted, the collars **47, 48** (after cooling to room ambient) can be moved along shaft **42** only by applying a push-off force of from about 1,500 to about 3,500 pounds (680 kg. to 1,590 kg.).

When the shaft stress is to be caused by the shrinking of collars thereon (or by the expansion of a previously cooled shaft against the bore of a room ambient collar), the amount of heating or cooling of one part relative to the other may be readily calculated by using the well known relationship: $(\Delta T)(D)(k)=\Delta D$; where k is the coefficient of thermal expansion of the material that is being heated or cooled, ΔT is the temperature difference (or gradient) under consideration, and ΔD is the change in diameter due to the change in temperature.

With reference now to FIGS. **1** and **4**, it is noted that, after final assembly of the motor **20**, the shaft **42** is received in a pair of sleeve bearings **38, 51** which are carried by hubs **52, 53** of end frames **24, 23** respectively. Also shown in FIG. **4** are parts of thrust systems which include thrust plates **54, 56**, and the thrust runners **43, 57**. It will be understood that the thrust runners have both a lubricant slinger portion and a thrust washer portion. Details of arrangements such as these are now available elsewhere (e.g., in commonly owned J. Stokke et al U.S. Pat. No. 3,793,543, which issued Feb. 19, 1954 and the entire disclosure of which is incorporated herein by reference), and detailed discussions pertaining to the subject matter disclosed in that patent are not repeated herein.

With reference to FIG. **4**, the spatial relationships between the shaft **42**, collars **47, 48**, rotor body **39**, and bearings **38, 51** should be noted. It will be appreciated that the rotor body **39** is secured to the shaft **42** between first and second ends **58, 59** thereof. Accessory members (in the form of collars **47, 48**) are positioned on the shaft so that they each extend axially from a first axial location on the shaft (adjacent a face **44** or **61** of the rotor body) toward an end of the shaft. The collars thus each extend to a respective second or third axial location on the shaft. Each collar has a surface region (e.g., face **46** of collar **47** against which the thrust runner **43** abuts) that is disposed adjacent the respective second (or third) axial location.

More specifically, such surface region is located so that it may be disposed closely proximate to a plane that is normal to the axis of the shaft **42** and passes through (or closely adjacent to) bearing means that support the shaft. For example, and as can be quickly ascertained from a review of FIG. **4**, the face **46** is closely proximate to a plane represented by the dashed line **63**. It is again emphasized that the shaft **42** is in a compressed (and therefore stressed) condition along the surface region

that is grasped by the collar **47**. In order to provide a frame of reference for "closely proximate", it is noted that the normal spacing between the face **46** and the end of the hub **52** (and therefore the bearing **38**) is only on the order of about one-eighth of an inch (3.17 mm) for motor **20**.

FIGS. **2** and **3** illustrate constructions that have been used long prior to the present invention. More specifically, FIG. **2** illustrates an arrangement wherein a rotor body **66** of conventional construction is heat shrunk onto a straight steel shaft **67** and two aluminum thrust collars **68, 69** (with bore dimensions as set out above for collar **47**) are heat shrunk onto the shaft **67**. The faces of the collars **68, 69** have been used to provide faces against which thrust runners substantially identical to the thrust runners **43** were pressed. FIG. **3** shows yet another prior approach wherein a rotor body **71** was heat shrunk onto steel shaft **72**. In this approach, aluminum collars **73, 74** were heat shrunk at predetermined positions onto the shaft **72** and provided faces against which thrust runners could be positioned prior to final assembly of a motor.

The straight shaft of FIG. **2** was one-half an inch, while the stepped shaft of FIG. **3** was five-eighths inch diameter in the central portion thereof, with one-half inch shaft extensions. The arrangement shown in FIG. **2** has been, and continues to be, utilized when the axial length of the rotor body **56** is relatively small. The approach represented by FIG. **3** was used when the rotor body was relatively long with an increased shaft length between bearings, and the enlarged portion of the shaft provided more rigidity as compared to a one-half inch straight shaft for the same application. It will be appreciated from a comparison of FIGS. **2** and **3** that a construction according to FIG. **3** would be more expensive to manufacture. This is not only because of the stepped shaft, but also because the laminations for rotor body **71** have a different shaft bore diameter than the laminations for rotor body **66**.

In FIG. **5**, an enlarged sectional view of the collar **47** has been illustrated. It will be noted that collar **47** may be quickly manufactured by boring a piece of round stock and then cutting a step therein with conventional machined tools.

FIG. **6** illustrates a prior art construction for a ball bearing motor wherein a rotor body **76** is heat shrunk onto a portion **77** of a stepped shaft **78**. The shaft portion **77** was three-quarters of an inch (19 mm) in diameter. Portions **79** and **81** of shaft **78** were machined to provide a one-half inch diameter shaft extension; and approximately 15 mm diameter seats **82, 83** were machined so that the inner races of commercially available 15 mm inner diameter ball bearings **87** could be pressed against the shoulders **84, 86**.

Reference is now made to FIG. **7** for a description of another embodiment of the present invention. More specifically, the rotor assembly **88** there shown may be used in lieu of the rotor assembly shown in FIG. **6** at a substantial savings in expense. For example, the rotor body **89** was made with laminations having the same shaft bore size as those used for the rotor bodies shown in FIGS. **2** and **4**. Accessory members, illustrated as collars **91** and shown in more detail in FIG. **9**, were heat shrunk onto a one-half inch diameter straight shaft **92**. The collars **91** were made from the same materials, and processed in substantially the same way as the collars illustrated in FIG. **4**; except that bearing seats **93** were machined on each collar so that the inner race of a bearing **87** could be pressed thereon in abutting relation with a shoulder or face **94**. Fully satisfactory results have been obtained when the seats **93** were turned (while the collars are on shaft **92**) to a diameter of from 0.5905 to 0.5908 inches and a bearing having a nominal inner diameter of 15 mm (0.5903 to 0.5906 inches) was pressed thereon. The precise axial location of the bearing seats **93** along shaft **92** are determined by precisely locating collars **91** on the shaft, all as is described in more detail hereinafter.

In FIG. **8**, still another embodiment of the present invention is illustrated wherein a rotor assembly **96** includes a rotor body **97**, a one-half inch diameter straight shaft **98**, a pulley end collar **99** and an opposite pulley end collar **101**. The collars **99** and **101** are formed of the same aluminum materials as the collars described hereinabove, and extend between axial positions substantially determined by rotor body end faces **102, 103**, and planes **104, 106** (which are substantially normal to the axis of shaft **98** and closely adjacent to axial ends of the bearings **87**).

It should now be appreciated that the collars **99** and **101** tightly grip and continuously stress the shaft **98** in predetermined regions thereof, substantially in the same manner and for the same reasons that collars **47** and **48** grip the shaft **42** (see FIG. **4**). The collars **99, 101**, and rotor body **97** are assembled with shaft **98** in generally the same manner as has been described above in connection with the other illustrated embodiments of my invention.

FIG. **10** is a flow chart of steps for one process that may be followed when making the illustrated embodiments. At the outset, it is noted that specific materials and nominal dimensions of parts have been identified in FIG. **10** only for purposes of disclosure and not necessarily for reasons of limitation. The referred to process includes selecting a desired size and type of material for use in making shafts. Thereafter, shafts of the desired length are cut from the stock material and chamfered at the ends. Then, in sequence, the shafts are straightened and washed, and ground to establish a desired concentricity; flats are provided on the shaft extensions (when desired); and a rust inhibiting treatment (such as phosphatizing) is performed. Collars which have previously been manufactured are then heated and shrunk onto the shaft, after which a rotor body is heated and shrunk onto the shaft. The second collar is then heated and shrunk onto the shaft.

If the ultimately resulting rotor assembly is to be utilized in a sleeve bearing motor, the appropriate collars will have been selected, and the rotor subassembly has thrust runners assembled therewith. However, in the case of rotor assemblies intended for ball bearing applications, the first and second collars are finish turned to provide a bearing seat or journal, and ball bearings are press-fit onto the journals before the resultant rotor assembly is advanced to final motor assembly.

In FIGS. **11** through **14** I have illustrated apparatus of one simplified and useful type which may be utilized to produce preferred embodiments of the invention. FIG. **11** schematically depicts a portion of the apparatus which is utilized during the assembly of firt collar **91** to shaft **92** when the assembly of FIG. **7** is to be constructed. Briefly, the fixture of FIG. **11** includes a base plate **107** with a guide ring **108** fastened thereto and a bottom plate **110** fastened thereto. Slidably received by the bottom plate is a stop block **111**. Stop blocks of various thicknesses may be used in the fixture so that the

axial distance between a top surface **112** thereof, and the surface **113** of the base may be varied in any predetermined manner.

Also included in the fixture of FIG. **11** are means for guiding a shaft during the assembly operation and for insuring that the shaft is maintained perpendicular relative to the surface **113**. Once such means for guiding is illustrated in the form of conventional vee-blocks **114** and a support **115** for the vee-blocks. The collar **91**, after being heated to a predetermined range (e.g., 400 to 700 F.), is positioned with a face **94** thereof engaging the surface **113**, and with the collar held in an aligned position by a guide ring **108**. Thereafter, the shaft **92** is hand placed through the bore of the collar until the shaft end **116** comes to rest against stop block **111**. A vee-block **114** is used to maintain the shaft **92** in centered relation with the collar **91** until the shaft has been seated against surface **112** of the stop block **111**. After a very short interval of time (e.g., 1 or 2 seconds), the collar **91** will have cooled sufficiently to be tightly locked to the shaft **92** whereupon the shaft and collar are positioned in a fixture that is schematically depicted by FIG. **12**.

The fixture **118** includes a base **119** which is provided with a hole **121** that will accept the shaft **92** with a slip fit, and insure that the shaft **92** is held perpendicular to a surface **122** of the fixture. Projecting from surface **122** is a removable spacer ring **123** which is selected to be of an axial length corresponding to a predetermined axial dimension that is to be established and maintained between the face **94** of collar **91** and face **124** of a rotor body **89**.

Having selected a ring **123** having a desired height, and positioned it (or suitably fastened it) relative to the fixture **118**, the shaft **92** with collar **91** shrunk thereon is positioned in the fixture. Thereafter, a preheated rotor body **89** is slipped down shaft **92** until the face **124** of the rotor body seats against the spacer ring **123**. The operation just described may readily be accomplished by hand, so long as caution is used and the preheated rotor body is handled with gloves or tongs.

FIG. **13** illustrates yet another part of the apparatus which is utilized to position a second heated collar **91** so that a face **94** thereof will be a predetermined desired axial distance from the face **94** of a collar that is resting against surface **126** of a machine base **127**. The apparatus of FIG. **13** includes not shown supporting structure which supports and upper frame member **128** in fixed relation to base **127**. Fastened to upper frame member **128** is a pneumatic cylinder **129** which is thus held a predetermined fixed distance from the upper surface **126** of the machine base **127**.

A rod **131**, which is driven by the cylinder **129**, is permitted to travel (upon actuation of the cylinder) until a stop nut **132** carried on the free end thereof bears against a spacer block **133**. The actual travel of piston rod **131** thus depends on the height of the particular spacer block **133** which has been selected. It will be understood that a spacer block **133** having a relatively small axial dimension will permit a relatively long rod travel. The apparatus of FIG. **13**, in actual practice, is inclined so that the axis of piston rod **131** is disposed approximately 45° to a horizontal plane. With the piston **131** completely retracted, the shaft **92** (having the rotor body **89** and lower collar **91** fastened thereto) is positioned in the fixture with the rotor body **89** held in centered relation by a large vee-block **136**. Moreover, the bearing seat **93** of the lower collar **91** is seated against a concentric guide surface **137** which is carried by the machine base. The vee-block and guide seat **137** cooperate to hold the shaft **92** perpendicular to the surface **126** of the machine base and in alignment with the piston rod **131**.

A heated collar **91** is then snapped into a collar holder **135** and cylinder **129** is actuated to slip the upper collar **91** to a desired final position along shaft **92** so that the faces **94** of the collars **91** are a predetermined desired distance apart. it should now be understood that the selection of a spacer block **133** (or plurality of the same) having a desired total axial height will insure that the upper spacer **91** is moved to a precise axial location along shaft **92** when the surface **94** of the lower collar is seated against the surface **126** of the base member. Since the preheated upper collar **91** will loosely slide along the shaft **92** while it is in a heated condition, it is necessary to provide some means for holding the collar to insure that it does not travel further than desired toward the end face **138** of rotor body **89**. Various means may be provided to frictionally engage the surface **93** of the collar **91**.

FIG. **14** clearly reveals that the collar holder is formed with a major portion thereof to conform to the configuration of collar surface **93**. The collar holder also has seated therein a pair of commercially available spring plunger pins **139**. For example, the pins **139** were manufactured by the Vlier Engineering Corporation, **8900** Santa Monica Boulevard, Los Angeles, California. The spring plunger pins are so located that they may be depressed to permit seating of collar **93** against the collar holder. Then, as the center of the collar moves beyond a line passing through the two plunger pins, the plungers move outwardly to frictionally grasp the collar. After a collar is shrunk on the shaft, the holder **135** withdraws and the pins slide along the collar. Holder **135** should be made of a mineral filled phenolic material or other suitable material to minimize heat loss from the heated collars.

The assembly of a collar with shaft **92** in the manner just described establishes a permanently stressed condition along first and second sections of the shaft which are substantially co-extensive with the collars. In the case of the structures shown in FIGS. **4** and **8**, a permanently stressed condition is established along first and second sections of the shaft. The first such section extends between first and second axial shaft locations; and the second such section extends between third and fourth axial locations. As will be appreciated from a review of FIGS. **4** and **8**, first and second faces of the rotor body are disposed substantially at the second and third axial locations, respectively. Moreover, after final assembly of a motor that includes the structure of FIGS. **4** and **8**, the first and fourth axial shaft locations will be proximate to an end of a bearing unit.

In the case of ball bearing units **87** (see FIG. **8**), the first and second locations are nearer to the outboard faces of ends of the bearing units and in the vicinity of planes **104, 106**. However, in the case of sleeve bearing units **38, 51** (see FIG. **4**), the first and fourth axial shaft locations are more proximate to the imboard axial ends or faces of the bearing units and in the vicinity of planes as indicated for example at **63**.

From tests that have now been performed on motors embodying the invention as illustrated herein, I have concluded that substantial savings can be realized in practice by using one-half inch straight shafts in many applications where larger diameter stepped shafts have been used heretofore. For example, in many applica-

tions where oversized stepped shafts have been provided because of the need to provide ball bearing seats and shoulders, use of the teachings presented herein can result in substantial cost savings without sacrifice in product quality. In applications where stepped shafts have been used to avoid noise problems, savings may be realized without undue degradation of product quality. Somewhat surprisingly, utilization of shrink fitted aluminum collars provides extremely good results.

In FIG. **15**, a number of points have been plotted based on test data that was taken on motors constructed according to the prior art and from motors that were constructed to embody different aspects of the present invention. A number of different motors, designated **A1, A2, B1, B2, C1, C2,** and **D1, D2** were each soft mounted on a test fixture and driven with an electromagnetic driver positioned in spaced relation to the motor shell. The electromagnetic driver was then excited at different frequencies by the amplified output of an oscillator. The motors were supported so as to be free to vibrate, and an accelerometer was attached to the shell of each motor under test. The output of the accelerometer was then fed into an oscilloscope so that the relative magnitudes of acceleration would be displayed on the vertical axis, and the output of the oscillator was fed into the oscilloscope's horizontal input. The oscilloscope then revealed the resonant frequency for the motor under test when the oscillator frequency was varied.

Prior to the tests just described, the deflection of the shaft for each motor was determined for a condition where each shaft would be supported at the bearing journal locations and a two hundred pound concentrated load would be applied to a point on the shaft midway between the two bearing journals. The resonant frequency (in Hz) and the shaft deflection (in mils, or thousandths of an inch) for each motor is plotted in FIG. **15**, and a curve has been fitted to the points so plotted. The points **A1, A2,** correspond to two different sleeve bearing motors made according to the prior art wherein a three-quarter inch stepped shaft (one-half inch shaft extensions) was utilized. Motor **A1** had a 4.82 inch long shell whereas motor **A2** had a 5.32 inch long shell.

Motors **B1** and **B2** used a one-half inch straight shaft, with motor **B1** having a 4.82 inch long shell, and motor **B2** having a 5.32 inch long shell. Motors **C1** and **C2** had 4.82 inch and 5.32 inch shells, respectively, and each had a straight one-half inch straight shaft with steel collars (otherwise similar to collars **47, 48** in FIG. **4**) heat shrunk thereon. Motors **D1** and **D2** has 4.82 inch and 5.32 inch shells respectively; one-half inch straight shafts, and aluminum collars shrunk thereon (as shown in FIG. **4**).

It will be recognized that motor **C2** represented measurable improvement over the prior art motor **B2**. A comparison of motors **C1** and **B1** suggested that some degradation of performance could result by using a steel collar. When data points **D2** and **C2**, are considered, it will be understood that straight shafts with aluminum collars may deflect more than similar shafts with steel collars. However, the resonant frequency of motors **D2** was significantly higher than that of motors **B2** and **C2**. The motor **D1** on the other hand had a higher resonant frequency and less shaft deflection, as compared to motors **B1** and **C1**.

Generally, a motor resonant frequency of about 600 Hz or greater means that a motor will normally have an acceptable noise level. Thus, the increase in resonant frequency for the motors **D1** and **D2** is an advantageous result. It is somewhat surprising, however, that the resonant frequency for the motors with aluminum collars was greater than the resonant frequency for the motors with steel collars (i.e., motors **C1** and **C2**). It is at least partly because of the data shown in FIG. **15** that it is presently believed that advantages are associated with maintaining a permanently stressed condition along predetermined unsupported lengths of a motor shaft, and that desirable increases in resonant frequency of such motors are apparently due to an increase in the spring stiffness of the overall motor system due to such stressed conditions. Moreover, it is believed that better results were achieved with aluminum collars than with steel collars because the aluminum collars (being relatively soft as compared with a steel shaft) conformed to shaft surface irregularities and caused a more uniform stressed condition. On the other hand, it is believed that when both the shaft and a steel collar are provided with concentric, regular surfaces, it should also be possible to attain a more uniform stressed condition.

While one preferred method of producing embodiments of the invention has been described, it will be understood that other mechanisms may be used to maintain prestressed shaft sections. Moreover, it may be desired to first secure a rotor body to a shaft and then stress predetermined shaft sections, e.g., by shrinking collars on the shaft substantially adjacent or proximate to each end of the rotor body. This sequence also may be followed, of course, when structures of the type shown in FIG. **7** are manufactured. Reference has just been made to "substantially" adjacent or proximate to a rotor body end face. In the context of laminated cores, this may actually mean that a collar will be within one or two lamination thicknesses from a rotor end face, since predetermined dimensions between precisely machined collars are easily maintained in my process, whereas a laminated core usually will vary plus or minus one lamination from a nominal length. Thus, if one collar abuts a rotor end face, the other collar may be as much as two lamination thicknesses (typically one lamination is 0.020 to 0.025 inches thick) from the opposite end face.

While a number of different embodiments have now been described, it should be undestood that modifications will readily suggest themselves to those of ordinary skill in the art. Moreover, different aspects of the invention may be combined together, or only some aspects of the presently disclosed inventions may be practiced. Accordingly, the scope of the present invention is to be measured only by that of the appended claims.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. A dynamoelectric machine including a stator, a pair of bearing units, a shaft supported by the bearing units and having two ends; a rotor body having first and second ends secured to a portion of the shaft between the ends of the shaft; and a pair of accessory members positioned on the shaft and each extending axially toward one end of the shaft from axial locations along the shaft at least substantially proximate to the first end of the rotor body, to other axial locations along the shaft; said accessory members each having a surface region at the end thereof nearest a respective end of the shaft disposed generally adjacent to a plane that is normal to the axis of the shaft and in proximity to an end

face of the respective bearing unit supporting the shaft; said shaft being in a substantially permanently stressed condition at least along the surface regions thereof that extend from an axial position at least substantially proximate to the ends of the rotor body to the other axial locations.

2. A rotor assembly comprising a shaft; a rotor body secured to the shaft and located between the ends thereof; a first collar secured to the shaft with an interference fit and located between a first end of the shaft and a first end of the rotor body; a second collar secured to the shaft with an interference fit and located between a second end of the shaft and a second end of the rotor body; said collars having machined bearing seats which are concentric with the shaft; a first ball bearing unit secured to the first collar; and a second ball bearing unit secured to the second collar; said first and second ball bearing units having an interference fit with a bearing seat of not substantially more than about 0.0005 of an inch.

3. The rotor assembly of claim 2 wherein the first and second collars are secured to said shaft with a shrink fit.

4. A dynamoelectric machine including a stator, at least one ball bearing unit, a shaft supported by the at least one ball bearing unit at first end thereof; a rotor body having first and second ends secured to a portion of the shaft between the ends of the shaft; and at least one accessory member positioned on the shaft and extending axially toward the first end of the shaft from a first axial location along the shaft between the first end of the rotor body and the first end of the shaft; said at least one accessory member having a machined bearing seat which is concentric with the shaft, said accessory member being secured to the shaft by an interference fit, and said at least one ball bearing being seated on said at least one accessory member, said at least one ball bearing having an interference fit with the bearing seat of not substantially more than about 0.0005 of an inch.

5. A rotor assembly for a dynamoelectric machine comprising a shaft having two ends; a rotor body, having first and second ends, secured to a portion of the shaft between the ends thereof; and two accessory members positioned on the shaft and each extending axially toward an end of the shaft from axial locations along the shaft that are at least substantially proximate to an end of the rotor body, to other axial locations along the shaft; said accessory members each having a surface region at the end thereof nearest an end of the shaft for disposition generally adjacent to a plane normal to the axis of the shaft and closely proximate to an end face of a bearing unit for supporting a shaft end; said shaft being in a substantially permanently stressed condition at least along the surface regions thereof that extend from an axial position proximate to the ends of the rotor body to the other axial locations.

6. The rotor assembly of claim 5 wherein the accessory members are steel collars.

7. The rotor assembly of claim 5 wherein the accessory members are secured to said shaft with an interference fit.

8. The rotor assembly of claim 7 wherein the accessory members are secured to said shaft with a shrink fit.

9. The rotor assembly of claim 8 wherein the accessory members are aluminum collars.

10. The rotor assembly of claim 8 wherein each accessory member establishes a seat for a ball bearing unit.

11. The rotor assembly of claim 10 wherein each accessory member is an aluminum collar.

12. A dynamoelectric machine including a stationary stator structure, a pair of bearing units supported by the stationary stator structure, a rotatable shaft having first and second ends supported for rotation by the bearing units and having two ends; a rotor body secured to the shaft for rotation therewith, with the rotor body having first and second ends spaced from the first and second shaft ends so that the rotor body is secured to the shaft between the ends of the shaft; and a pair of accessory members positioned on the shaft with each extending, from closely proximate to the rotor body, axially toward a respective end of the shaft; said accessory members each having a surface region at the end thereof nearest each respective end of the shaft that is disposed adjacent to a plane normal to the shaft axis and that is disposed in close proximity to an end face of the respective bearing unit supporting the shaft; said shaft being in a substantially permanently stressed condition at least along the surface regions thereof that extend from an axial position at least substantially proximate to the ends of the rotor body to the above-mentioned planes in close proximity to each bearing end face, with the stressed condition bearing maintained by the accessory members.

13. A rotor assembly comprising a shaft; a rotor body secured to the shaft and located between the ends thereof; a first collar secured to the shaft with an interference fit and located between a first end of the shaft and a first end of the rotor body, said first collar having a first end disposed closely proximate to the first end of the rotor body; a second collar secured to the shaft with an interference fit and located between a second end of the shaft and a second end of the rotor body, said second collar having a first end disposed closely proximate to the second end of the rotor body; said collars having machined bearing seats concentric with the shaft at the second ends thereof which are also most proximate to each respective shaft end; a first ball bearing unit secured to the first collar on the bearing seat thereof; and a second ball bearing unit secured to the second collar on the bearing seat thereof; said first and second ball bearing units having an interference fit with the respective bearing seats of not substantially more than about 0.0005 of an inch; and wherein said first and second collars constitute means for strengthening the shaft between the ends of the rotor body and bearings whereby shafts of predetermined relatively small diameters may be utilized.

14. A rotor assembly for a dynamoelectric machine comprising a shaft having two ends; two bearing units; a rotor body, having first and second ends, secured to a portion of the shaft between the ends thereof; and two accessory members positioned on the shaft and each extending axially toward an end of the shaft from locations substantially proximate to an end of the rotor body, to other respective axial locations more proximate to an end of the shaft; said accessory members each having a surface region at the end thereof nearest the respective shaft end disposed adjacent to a plane normal to the shaft axis and closely proximate to an end face of a respective bearing unit; said shaft being in a permanently stressed condition along the surface regions thereof that extend from proximate to the rotor body ends to the bearing units.

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